Remarks:

In response to your review of **Claim 17** as not patentable under 35 USC 103 (a) a basis of obviousness: Patent 5,081,832 (Mowill) describes at least two a gas turbine engine prime mover configurations having two non concentric rotor spools of which one rotor spool supplies mechanical output power driving means. Each spool has a compressor rotor and turbine rotor. The output power spool shaft (12) has an external load drive connecting means (18) at one end to drive the external load (20). Load (20) is case attached to the engine body (10) and shaft (12) drives the external load (20) thru connection (18) receiving the mechanical output power of the spool shaft assembly (12). Loads to be driven thru connection (18) could include a gearbox apparatus, clutches, etc with related shafting. Load (20) could include a high speed electrical generator without a gearbox via the load drive means(18) (an end drive coupling to shaft 12). Furthermore this Mowill envisioned high speed alternator with load drive 18 (an external drive coupling means) yields an alternator rotor with its own rotor bearings (like Yoshida, 4,564,778 generator / motor) and not like 10/809,719 where the alternator is integrated into the power output spool rotor and thus have a common bearing arrangement.

The Yoshida patent (4,564,778) describes a number of generator and motor system configurations not requiring rotor contact brushes. Magnets 40 are mounted N-S radially on the OD of a rotor core of which configures an axial stack of iron laminats 14. Although this arrangement of the magnets and rotor laminat may be suitable for the application of relative low speeds, it would not be suitable in the high speed

applications of the 10/809,719 application. The laminat rotor core not only would be difficult to assemble and balance but would have inherent rotor radial movement in a high speed alternator application causing rotor dynamic stability issues. A radial laminat core shift (due to centrifugal force movement) as little as .002 would yield a rotor dynamic instability. Both Mowill and Yoshida patents in this correspondence require an external drive coupling to load drive the alternator / generator.

10/809,719 figure 4 detail 144 depicts a (non laminat core) solid core alternator rotor located between the engine #2 rotor spool bearings 125 and 186. As a note alternator rotor magnets of 10/809,719 radially have a low radial profile to minimize centrifugal forces, are positioned axially n-s and are retained by an OD sleeve retentions means of which Yoshida does not. Mowill patent notes a possible externally load driven high speed generator system (column 3 line 21-23) as noted - the load 20 (stator and rotor) is externally mounted to the engine body (10) where the spool shaft assembly (12) incorporates an external load drive 18 means to drive the external load (20).

Furthermore all power is from the output power spool drive external loads. Mowill patent column 1, line 51 – 53 “Means are provided connected to the shaft means for driving an external load, -----“

10/809,719 application integrates the alternator rotor 144 (the load) to the #2 rotor power spool 120 assembly, shaft 122 and use common rotor spool shaft bearings 125 and 186 yielding simplicity and lower cost by removing the complication and rotor dynamic complexity of an external drive - load system.

By current industry definition a high speed alternator (AC electrical power generator)

system is used primarily in conjunction with gas turbine engines (running at the same rotor high speeds) allowing non synchronous operation with electronics to convert the high frequency. Also the high speed alternator being much lighter than conventional power plant electrical generators of synchronous speed operation allows quicker engine rotor response. Other applications include piston engines and auxiliary power systems with attachments to gearboxes to attain high speed operation. Other characteristics of high speed alternators include: brushless rotor shaft contact, a solid iron base rotor core with permanent magnets integrated and a outer sleeve to retain the magnet to the core under high rotational shaft speed centrifugal loads. Mowill mentions possible external load 20 of a high speed generator and a drive means 18 connected to the engine rotor power spool shaft 12; this would require independent alternator / generator rotor shaft. Mowill writes of reduced power rotor (HP spool) speeds col. 4, lines 3-6 for improved efficiency at low power but his concept turbine rotor spool system, external drive means and the external alternator system may not be suitable for all operational rotor speeds – not reduced to practice. 10/809,719 overhung rotor spools (page 3 col. 2, lines 8,9,10 and figures 1, 2, 3 and 4)with and without the alternator have been reduced to practice and operate at any rotor speed.

(HP spool)Yoshida patent **4,564,778** has a brushless motor or generator but does not have the high speed rotor technology of an outer sleeve retention mean to hold the magnets to the alternator rotor core. Also the laminated rotor core would not be suitable for high speeds.

In summary neither the Mowill patent 5,081,832 or Yoshida patent 4,564,778 with their

independent generator / alternator and corresponding alternator rotor shaft bearings require an external load drive means - do not teach the high speed alternator / generator technology or integration to the #2 spool power output rotor as defined is application 10/809,719 that yields simplicity and lower cost.

#17 Claim Amended supporting remarks

This claim with engine test data represents an improvement over the short comings of the prior art as follows:

The **6,314,717 patent** a single spool rotor microturbine with an integral alternator rotor operates at maximum RPM for maximum power electrical output power and reduced output power conditions. In an effort to reduce the fuel consumption for lower output power requirements, tests were conducted at reduced rotor spool speeds and subsequent reduced engine air flow. The combustor, having multiple fixed fuel supply nozzle configurations were designed for output power at maximum rotor design speed and emissions. A maximum power turbine exhaust gas temperature was set at ~1350F with consideration of turbine and nozzle materials no cooling means recuperator materials at maximum power conditions. At this reduced power testing, having reduced rotor spool speed, the engine had limited output power due to the fixed combustor geometry and non variable fuel nozzle quantity (no staging) trying to maintain low emissions species and avoid a carbon building due to excessively rich F/A mixture combustion. The resultant F/A richness increased with power yielding a carbon issue (enough to cause primary zone blockage), high CO and UHC species. On a positive note the engine design at constant maximum rpm exhibited good low emissions and instantaneous power output response at

any power level without rpm loss (loading and unloading). The rotor spool assembly design incorporated a 2 bearings straddled mounted configuration about the alternator rotor and common to the power spool rotor with a resultant overhung aft positioned radial compressor and turbine rotors. One engine configuration incorporated a typical weighty exhaust gas heat exchanger for improved cycle efficiency but at high power off loading (high stored heat energy) to a no load condition, combustion flame extinctions becomes at times an issue. The other version of this microturbine is a simple cycle without a heat exchanger exhibits typical low efficiency, increase fuel usage but no off loading issues. Both engine configuration have rich F/A mixture issues at reduced rotor speeds creating carbon build up in the combustor. Variable geometry as an air flow control mechanism could be incorporated to the combustor primary flame zone for off design operation allowing more air to maintain an optimized primary zone F/A mixture for low emissions combustion and no carbon issues, but adds complexity and further expense. 10/809,719 application rids these short-comings.

The Capstone turbine patents **# 5,497,615 and 5,685,156,** represent a microturbine having a (air bearings) three bearing single rotor power spool system with an alternator incorporated. The spool rotor assembly is considered semi-flexible (no drive shaft). The engine combustor incorporates staged fuel supply premixing fuel nozzles to address off design power operations and low emissions at maximum rotor speed (~constant engine air flow). Reduced rotor speed operations for reduced power requirements are limited; although the range of operation is improved over 6,314,717 patent the F/A mixture control for low emissions is limited and the rotor dynamic critical shaft speeds using air

bearing need to be avoided (pass thru quickly). During engine off-loads from high power conditions a combustor case blow off valve is incorporated to prevent combustion flame-extinction during the time needed to dissipate the heat exchanger heat energy. The alternator rotor has a bearing straddle mounted system with shared support to the adjacent compressor rotor; and between the compressor rotor and radial turbine rotor another air bearing is located leaving the turbine rotor over hung. Like the 6314717 reducing the spool rotor RPM for improved fuel economy is limited in power range to avoid issues of emissions and or combustor carbon generation.

The **Mowill patent 5,081,832** represents a two spool rotor system where the low power compressor (LPC) rotor spool has an external drive means to a gear box which in turn drive a synchronous generator. Also contemplated, without a gear box, the external drive means can be coupled to an alternator rotor for non-synchronous electrical power generation. The LPC spool having a compressor and turbine rotor, pictorially exhibits a straddle mounted system (the aft bearing is not shown but is typically of industry in the exiting exhaust duct aft of the turbine). The LPC spool rotor bearing arrangement exhibited does not lend to the integration of an alternator rotor. An alternator rotor (outside of the engine body) with its own bearings (gearbox is not necessary) and is externally driven by the LPC rotor spool and has its own separate straddle mounted bearing configured all-of- which adds to complexity and cost over the patent application 10/809,719. Furthermore Mowill goes into detail design analysis of 2 spool rotor systems that theoretically delivers higher than 12:1 pressure ratio (even approach 20:1 of which will give higher cycle efficiency); but from authors of 10/809,719 microturbine design and test experience ~1875F is a limit (~11:1 pressure ratio), beyond will need to

incorporate a turbine nozzle cooling means to the metal used, thus adding to complexity and added cost. Mowill's sketches eludes to a power turbine nozzle cooling means capability.

Patent application 10/809,719 integrates an alternator rotor to the low power compressor rotor spool yielding simplicity over the **5081832 Mowill patent,** the latter of which incorporates an externally drive means for a generator, alternator or for other mechanical power applications. The electrical output power rotor spool of 10/809,719 is similar to the 6314717 (~ 4:1 compressor pressure ratio) but has a greater power output capability by using a lower compressor pressure ratio (~2:1) thus allowing more power from the turbine to drive the integral alternator/power rotor spool. The addition of the high power compressor turbo stage second spool rotor in this engine configuration over comes the short comings of the prior art microturbine –allowing for reduced output power rotor spool speed for off design / low power requirements yielding improved fuel economy thru reduced power needs to drive the low pressure ratio compressor. The initial engine test (this patent application / technology has been reduced to practice) results showed a primary zone F/A mixture ratio (without a variable geometry combustor) remaining fairly constant and exhibiting ~1.25 to 1.45 equivalent ratio range under various load conditions without any carbon issues or compromise in emissions thru reduced output power rotor spool speeds (as low as 55% of design speed). Equivalence Ratio = F/A (actual) / F/A (stoichiometric) 10/809,719 two spool engine test results are referenced the supplied **Power vs Fuel Flow curve.** For a given power requirement the output power rotor spool speed will have a unique rpm lower speed for lower power leaving the turbo rotor to respond for airflow

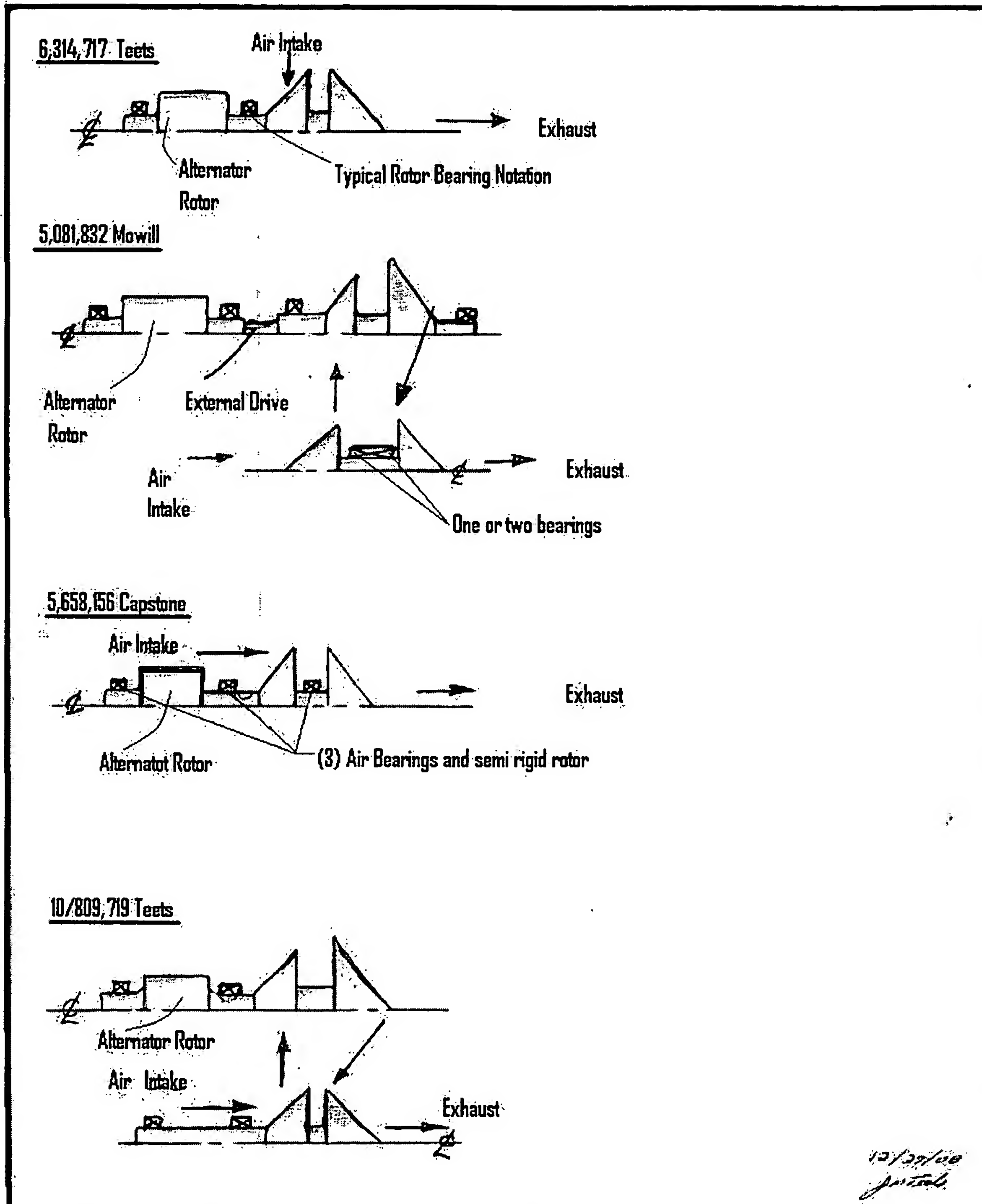
needs with the power spool turbine exhaust gas temperature limiting the speed range. With reduced power rotor spool speed capability fuel savings is recognized for part power applications.

Holding the output power rotor spool speed at a selected constant RPM reduced power level requirement, as the fuel flow is increased for increased electrical power output demand, the turbo rotor spool speed increases with the increase of exhaust gas temperature (EGT) of the power rotor supply to the turbo turbine , supplying more air flow to the combustor resulting in primary zone ~ constant F/A ratio with no carbon issue or emissions compromised (without variable geometry). This overcomes the need for the complexity of staged fuel supply nozzles of patent 5,685,156 and or the need of variable geometry in the combustor liner geometry as means to avoid a richening primary zone and subsequent carbon issues thus (equivalence ratio > ~1.6) simply maintaining a desire F/A mixture for emissions consideration. These test results were a first in the microturbine business allowing ~constant F/A mixture combustion at reduced power rotor speed operation for reduced power requirements with resultant reduced fuel flow as compared to maximum power spool rotor and speed at reduced power requirements.

10/809,719 engine application initial design selection of ~11:1 pressure ratio selection was based on not to exceed 1875F turbine inlet temperature and thus no turbine nozzle cooling was required. This engine design was for a prime mover application use in an average car with 70Kw (maximum power) considering acceptable acceleration, and at power requirements <10Kw (~80% of the time of typical vehicular use <60MPH) yields better fuel efficiency than existing vehicular engines. Reference the supplied curve "Engine Configuration vs Efficiency" that depicts the TMA Power, LLC (10/809,719) test data year

2005 of the TMA-70 (70Kw) two spool microturbine as compared to other typical engines on the market. Also as a note the 2 spool microturbine (10/809,719) using diesel fuel exhibited low emissions (less than tier 3 EPA limits) at all power levels and far superior than today's diesel engine which require post combustion exhaust treatments

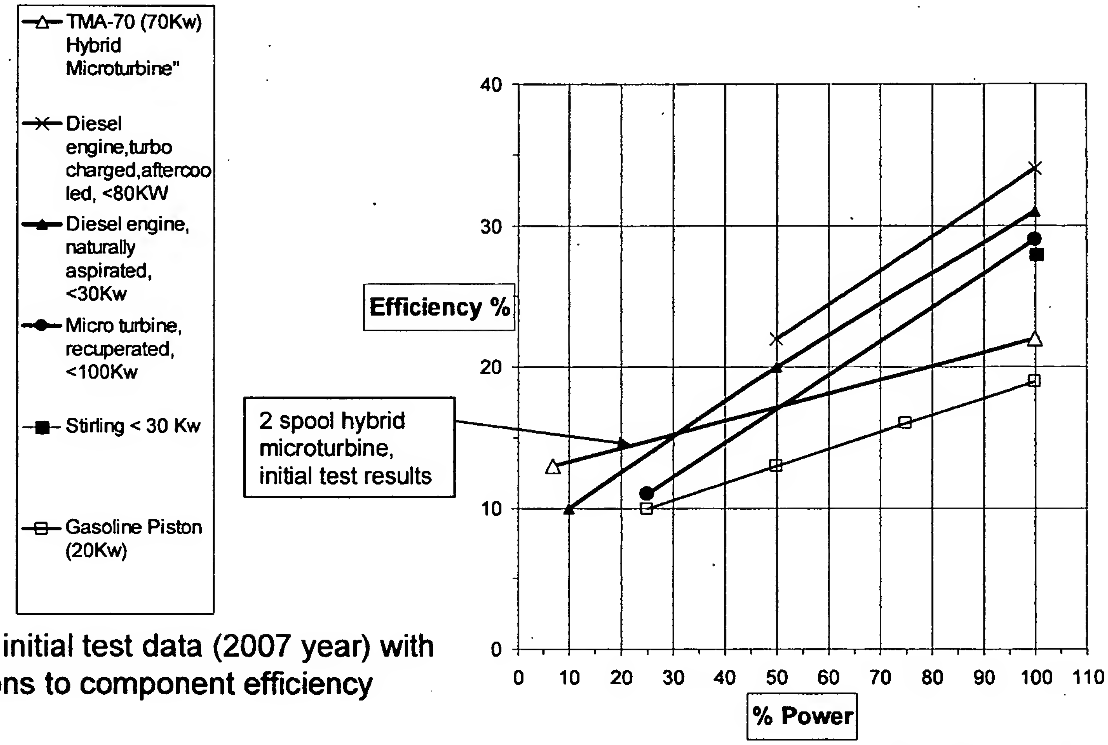
This patent application overcomes the engine rotor complexity of the Mowill 5081832 patent scheme of an externally driven high speed alternator rotor. Also the power rotor spool with the straddle mounted bearing scheme about the alternator integration yields a simple overhung radial compressor and turbine arrangement offering uniform rotor mass balance over that of 5081832 (improves rotor stability thru critical rotor speeds along with the bearings in a cooler environment improves durability). As a further note (10/809,719) the turbo-rotor spool (HPC) geometry (compressor rotor, turbine rotor and shaft) and the power rotor spool (LPC) use common core castings for cost considerations where-as Mowill exhibits different rotor geometries / configurations between the first and second rotor spools.



Overview of Prior Art compared to Patent App. 10/809,719

TMA Power, LLC

Engine Configuration vs Efficiency



TMA-70 initial test data (2007 year) with corrections to component efficiency

10/809,719 Patent Application

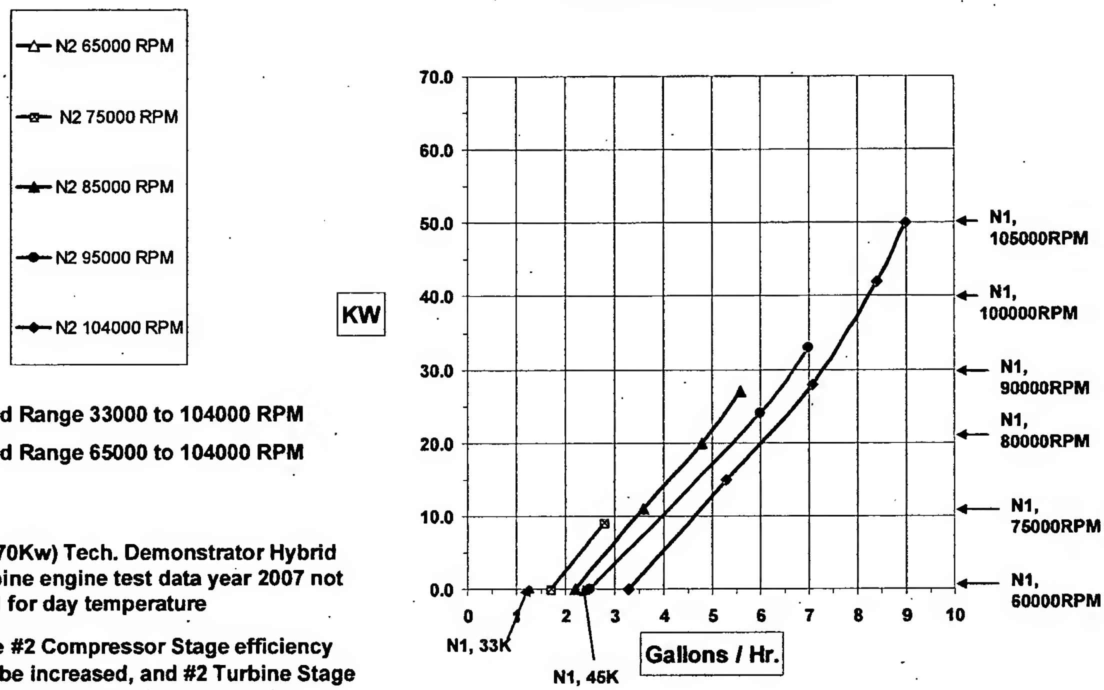
Preliminary

Definition of a microturbine: A gas turbine engine with < 500Kw non-synchronous electrical output power capability using a high speed alternator integrated to a output power spool rotor having a compressor rotor and turbine rotor.

The **Power vs Fuel Flow** curve represents 10/809,719 application number engine test data exhibiting reduced power needs at reduced turbo rotor spool and reduced power rotor spool speeds to allow improve fuel economy at off design conditions. Example: if the need is 10Kw, run the N2 power rotor spool speed at ~ 75000 rpm instead of 100,000rpm thus saving ~ 1.5 gal/hr. The N1 rotor spool or turbo rotor responds to the power rotor exit exhaust gas temperature driving the necessary air flow for the require output power. The exhaust gas temperature limit is set to prevent any excess temperature to the power turbine nozzle and or turbine blades and will dictate the N2 rotor speed required for a given power output. The end of each generated curve represents maximum turbine temperature limits of this technology demonstrator engine and will vary depending on turbine material temperature limits and or cooling scheme if incorporated.

TMA Power, LLC   Power vs Fuel Flow

10/809,719 Application



- N1 Speed Range 33000 to 104000 RPM
  N2 Speed Range 65000 to 104000 RPM
- TMA-70 (70Kw) Tech. Demonstrator Hybrid Microturbine engine test data year 2007 not corrected for day temperature
- Prototype #2 Compressor Stage efficiency needs to be increased, and #2 Turbine Stage pressure ratio needs to be increased

Preliminary

In response to your **review of claim 18** as not patentable under USC 103(a) which forms the basis of obviousness I submit the following: Mowill patent 5,081,832 duct #42 exhibits a fixed geometry duct diffusion means (gain more static pressure) – an axial expanding area flow area downstream and adjacent to the 1st stage compressor exit scroll #48 compressor. The duct #42 also transitions the #1 compressor discharge air flow to duct 46 and then on to the plenum #48 adjacent to the

the inlet of the #2 compressor (14). Attached to duct 46 is an air bleed valve 52 to help balance the airflow between the compressors (preventing a possible compressor surge) by

discharging a % of air flow from the #1 compressor overboard during transient engine operation. The air bleed valve 52 system is typical in industry of gas turbine engine having 2 or more rotor spools. Application 10/809,719 incorporates a variable position plate-valve 152, airflow 53 control means between the #1 compressor exit 32 and the #2 compressor inlet 168 as an internal air flow control means. The valve plate 152 in position 152A acts as an air flow restrictor to help balance the off design engine transient operation air flow to the #2 compressor preventing possible surge. Using the variable position valve plate 152 within duct 150 in a restricting position152A, the air flow 53 out of the #1 compressor during transient operation is reduced, yielding an increase in air flow pressure and subsequently reduced the #1 compressor rotational speed.

In response to your review of **claim 19** you stated that Mowill teaches various features as presented in my application 10/807,719. Mowill (HP spool, Fig. 1 and 2) does not teach an alternator rotor with permanent magnets integration to the #2 rotor spool but instead Mowill has a external load 20 that could be a high speed generator and is attached external to the engine body 12 having a drive means 18. Both Mowill and Yoshida electrical generators have external driving means and have their own dedicated separate bearings and seal systems. Application 10/807,719 integrates the alternator / generator to the prime mover #2 rotor spool as a driving means.

Strass patent notes (col. 6, lines 57-63) labyrinth seals can be used as sealing / isolation means between bearing compartments and gas chambers in turbomachinery. The turbomachinery compressor system has a case / body and a rotor with rotor bearing (26) (25)

and seals (28) (27). The compressor system/case having a separate body is externally coupled to a transmission system/case with its own brgs. and seals and intern also coupled externally to a generator/system having bearings and seals all of which are typical to industry. Also for ease of maintenance bearing and seals of compressor system/case are configured to the case such as to allow access for maintenance without rotor - case disassembly. The generators in Strass and Mowill patents are not integral to the engine output power rotor spool as in 10/809,719 but instead are located on a separate rotors within a separate body / case having separate brgs. and seals; requiring an external driving means. Strass notes lab seals of use in the configuration of compressor rotor – bearing – bearing housing – seals are not insertable with the #2 power rotor spool assembly as noted in my original claim 20. The Strass bearing Fig. 3, #26 and seal arrangement are axially insertable to the shaft #2 from outboard ends and into inner cavities 18c or 23. Also for ease of maintenance of seal and brgs. they can be removed from the compressor body without removing the compressor rotor 4 from the compressor body.

Mowill and Strass patent ideas, like other turbomachinery, use bearing and seals including labyrinth type but do not teach the 10/809,517 application module configuration #2 power rotor spool (incorporating integral alternator) with brg. and seal housing assembly as a module, insert-ability into the engine body passing and thru a compressor stationary shroud. I have canceled **Claim 20,** but integrated the rotor insert ability idea of the prior Claim 20 into **claim 19 (currently amended).** The informalities of claim 19 have been corrected. As an added note the idea of module having the alternator rotor assembled to the #2 rotor spool along with the noted brg. housing allows for an total rotor balance avoiding post rotor

component assembly to the engine body of less balance integrity. Specification Page 11, line 18 thru 21.

In response to your review of **claim 21** as taught by Singh. Oil squeeze film damping cavity 20 between the bearing outer race 13 and the respective inner bore of the bearing housing 16 is not new. Singh investigates improved seal ring systems for a bearing damper. The bearing housing 16 is typically hard mounted to the engine body as seen in this patent 5,085,521 Singh. But Singh incorporates unique cross sectional seal 30 shapes within unique receiving ring grooves 33 34 in bearing housing 16 to improve on the seal means (minimizing side leakage). Also unique in this patent are adjustable screws to limit the amount of radial movement of the bearing outer race within the oil damper operation. (col. 2, lines 33-38) denotes a shaft 15 with bearing 11 inner race 12 mounted incorporates the use of oil film squeeze damper cavity 20 located between the bearing assembly 11, outer race 13 OD and inner housing 16 (fig. 2) bore having outboard oil supply mean 39. 10/809,719 application reflects the oil squeeze film damping of the bearing-seal housing 126 of fig. 5 and engine housing 140, pg 29, line 1 thru 4.

In response to your review of **Claims 22 , claim 22 has been cancelled**

In response to your review of **Claim 23,** the Kawakami patent 5,246,352 a number of bearing arrangements schemes are presented. There are up to three radially displaced **oil squeeze film effect surfaces** created in the turbo charger radially displaced rotor bearings for rotor vibration dampening consideration. The innermost bearing - bearing journal bearing ID

surface co-acts with the rotor shaft and the outer most bearing -bearing journal OD co-acts with the housing ID. The radial series arranged bearing have radially displaced bearing journals with oil supplies for lubrication and in turn by nature these rotating journal surface area generate an oil film and act as **oil squeeze film effect** damper. Oil exits the journal bearing surface axially having no seals. The **oil squeeze film effect** areas (these noted areas do not have any axial sealing means to allow for the pressurizing of the oil about the cylindrical surface for an oil squeeze damper to be operate and may do well for that application but are not oil squeeze film dampers and are not suitable for the 10/809,719 application . Oil squeeze film dampers require a defined (axial and radial) pressurized annular cavity to function and incorporate axial end sealing means with non rotating surfaces. The patent 5,246,352 does not teach 10/809,719 Claim 23, reference Page 11, line 17-21, pg 12, line 5 -9 and page 27 line 5 thu 7 f the specification. As a note in the Kawakami patent the rotating sleeve bearings with developed surface oil film will be disrupted if an adverse radial load stops the bearing rotation.

In response to your review of **claim 24** as Mowill teaches a preswirl. In Mowill Figure 1 (closest in detail, is scant in detail) of 5,081,832 patent, shows a diffusion duct 41 that connects to conduit 46 and in turn attaches ~ 90 degrees to downstream air flow supply cavity adjacent to the 14 compressor entrance. This would infer that the air at this junction travels radially inward toward the compressor 14 axis centerline. Further review of Mowill specification, column 4 line 34, 35 the area cavity just before the compressor 14. is identified as 48 and is referred to as a plenum. A plenum in my experience is a fluid

settling zone not yielding direction of flow or inducing any compressor 14 air inlet preswirl. Application 10/809/719 specification page 30, line 10 – 16 and line 20 – 21 supports my **claim 24 (newly amended)** having a unique method of inducing air inlet preswirl thru a ducting means. Figures #11 and #9 depict the induced tangent radial in-flow 170 of the air entering the #2 compressor inlet. Also included in the claim 24 is your requested antecedent corrections "informalitie"changes.

In response to your review of **Claim 25** as not patentable in view of Mowill teachings per column 1 lines 52-59; Mowill writes of a internal combustion process within a engine contained combustor as a heat source means for a gas turbine application. The #2 compressor discharge air is mixed with internally supplied fuel within the engine housed combustor and combusted for hot energy gas to drive the #2 turbine rotor. Application 10/809,719 specification page 30, line23, 24 and page 25, line 1 states "Furthermore an external heat source could be incorporated removing the need of internal combustor". Thus as another means to supply heat energy to drive the #2 turbine rotor of the noted two spool system air from the #2 compressor discharge air would be heated thru a conventional heat exchanger where the external heating means heat energy is transferred thru the walls of the heat exchanger yielding heated air to drive the turbine. This would replace the heat energy source hot internal combusted gas conventional heating means to drive the #2 turbine. I have submitted **claim 25 (newly amended)** with clarification for your further review and consideration.

In response to your review of **Claim 26,** Kawakami patent 5,246,352 incorporates o-rings 33

as an oil sealing means between the bearing and housing. Although the o-ring material is resilient by nature, the bearing relative response to the shaft will be dictated by the stiffer oil film fluid forces leaving the o-ring only for sealing purposes. In 10/809,719 application the o-rings in fig.12, # 128 are incorporated to the seal 132 OD not only to act as an air seal means against cavity 168 but most important as a means to minimize the seal 132 wear with the potential relative to the shaft 122 radial movements during rotor operation. The resiliency of the o-rings would allow the seal to move radially during rotor excursions away from the shaft surface. Thus the Kawakami patent does not teach that technology of Claim 26.

In response to your review of **claim 27**, Claim 27 is canceled.

In response to your claim 28 review, I am submitting **Claim 28 (currently amended)** with more specifics. Referring now to Fig.16 the buffer air is supplied to the labyrinth seal 56 axially central from the back of the compressor rotor 42 via channel 70a, radially inward to a rotor centerline hole (with radial stress considerations) then passes forward to general radial hole 46 and on to the lab seal for buffering to keep the bearing oil from interring the lower pressure cavity compressor inlet 24. As a note the compressor rotor 42 is welded to the compressor shaft 42 so a hole installation can be configured prior to welding. Mason, a typical lab seal arrangement considers means to reduce windage losses thru the lab seal between compartments 40 and 41 and also the lab seal here is a means to reduce compartmental air flow across the seal. Furthermore the rotating knife edge seal has rotor dynamics issues over that noted in 10/809,719 application where the knife edge is static. The feeding of buffer air to the lab seal thru the shaft and centrally to the lab

seal axially not only adds in simplification of the air supply but prevent air fluid flow between compartments 24, 62 of fig. 16. pg. 24, line11 thru 13 Accordingly I have added further definition to the claim 28 now **claim 28(currently amended).**

In response to your review of Claims 29, 30,31and 32, I have **canceled claim 29, 30, 31 and 32.**

In response to your review of **claim 33**, see my response notes on claim 17 and also furthermore the Shekleton, 5343690 defines a method of impinging high pyrotechnic gas energy 130 and a nozzle arrangement 134 to rotate the gas turbine to ~40% rotor speed(figure 3, 140) where combustion is initiated, and with further pyrotechnic gas energy the system becomes self sustaining 141 and the start pyrotechnic gas is stopped. At this ~40% rpm (fig 3, 140) point the engine compressor delivers adequate air for the combustor for a F/A mixing process and together with the fuel injection and subsequent mixing combustion begin with stable flame energy to drive the turbine wheel. 10/809,517 application uses supplement air for impingement onto the compressor exit wheel blade area for the spool rotor rotation and also at the same time this downstream exiting compressor start air is used in the combustion process (fuel air mixing within the combustor) allowing low rotor speed starting, at ~10% speed prior to having developed compressor air flow supply. Page 34 lines 1 thru 15 and page 13, lines, 5 thru 13. **Claim 33 (currently amended)** I have added words in this two spool microturbine to reflect the air of the air start system is also used for early combustion process prior to compressor developed air flow for combustion F/A process.

In response to your **Claim 34** review, I have **canceled claim 34** as it relates to a single spool microturbine.

In response to your review of **Claim 35**, my main intent is to have coverage on the method of a rotor spool retention means specifically #1 spool rotor assembly 40 having a retention means 74 with a threaded end that co-acts with case 20 internal threaded end. Figure 6 pictorially shows the thread relationship between item 74 OD end thread and the receiving case 20 ID thread end. Per telecom you conveyed that an added thread note to the specification would be acceptable and accordingly Pg14 of specification line 2, pg. 20 line 15 **(threaded)** has been added. Also reference my notes supplied in this document concerning claim 17, on Mowill / Yoshida. The Straus patent, item 18c may be considered a bearing retainer sleeve but it refers to a axial case insertion from the outside end opposite the direction of the compressor rotor installations and does not teaching the 10/809,719 rotor spool 40 (with bearing and sleeve system) axially insertable from one end. This design choice allows for the rotor spool 40 retention means per claim 34 and as an assembly the bearing system – sleeve 54 could allow for balancing without subsequent disassembly for insertion into case 20. Accordingly I have supplied a revised claim 35, **Claim 35, (currently amended).**

In response to your review of claims 36 and 37. I have modified the claim 36, **Claim 36 (currently amended).** First note, an error on my part, the dependent claim 34 should have been 35. Also reference my claim 23 comments on Kawakami teachings. Furthermore on Kawakami teaches turbocharger rotor schemes having multiple rotating

sleeve bearings and oil film journal bearing surface(up to 3 journals in figures 1 thru 7) of the rotating bearings. Some rotor vibration dampening capability is mentioned thru these oil film journal surfaces as to have oil squeeze film effect. Figure 8, 9 and 10 depict a detail 26 OD with a gap 27 to the case housing ID of which could act as an oil squeeze damper area with related sealing means 81 for a oil squeeze damper to operate. For an oil squeeze film damper one needs to incorporate in the hardware: 1/ controlled small radial gap, a unique defined axial length for spring rate stiffness consideration and 3/ have shaft seals to control the oil fluid. The patent of Kawakami does not teach multiple oil squeeze dampers and at best pictorially show the potential of one area, with a rotating inner sleeve bearing.

I have changed claim 36 to incorporate the ideas of claim 37 (**claim 37, canceled**), as to have radially multiple oil squeeze film dampers capabilities.

Reference: Page 20 of specifications, line 15 thru 22, and pg. 21, lines 1 thru 3

Referring now to **Claim 35 Currently Amended,** per our telecom you allowed me to incorporated words is the specification that would clarify figure 6 and allow claim usage concerning threads on detail 74 and receiving inner thread of case 20. On page 26 of this document line 15, I added rotor **threaded** retainer, and on line 16, I added after 20 **having receiving inner thread of retainer 74.**

Also you had requested that I remove 122A and 63A, of pg 27 from the amendment filed 4/18/08. I believe base on my revised drawings filed 4/18/08 that 122A and 63A are correct and should remain.

In view of the foregoing, it is believed the currently amended claims (previously rejected) are patentable over the noted prior art and also that other claims 18, 21, and 23 previously rejected are patentable over the cited prior art in view of my remarks.

I would appreciate any assistance from you on possible claim rewording if necessary.

Reconsideration of the rejections of claim 18, 21 and 23 is respectfully requested.

Sincerely,

J. Michael Teets,

Signature:__________________________, Date:_______________